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LIQUID-COOLING TECHNOLOGY FOR GAS TURBINES
REVIEW AND STATUS

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ABSTRACT

A review of research related to liquid cooling of gas turbines has been conducted and an assessment of the state-of-the-art and future requirements has been made. Various methods of liquid cooling turbines are reviewed. Examples and results with test and demonstrator turbines utilizing these methods along with the advantages and disadvantages of the various methods are discussed. The more significant research, design data, correlations, and analytical methods are mentioned and voids in technology are summarized.

THE MAJOR ADVANCEMENTS and applications in cooled gas turbine engines to date have been with direct air cooling of the hot section parts. This has been due to the dominant influence of aircraft gas turbines wherein air cooling has been more practical. Mechanical complexities associated with liquid cooling systems along with inadequate heat transfer and coolant flow data have contributed to the lag in its application to gas turbines.

The main advantage of liquid-cooling is the potential to maintain low metal temperatures with high gas temperatures. This could be important to turbine hot section life particularly by corrosion-erosion, if the heavier, ash-bearing fuels are used.

The recent concern with our limited petroleum supply and the desire to reduce fuel consumption have rekindled interest in liquid cooling which will permit higher maximum gas temperatures resulting in greater cycle efficiencies and higher specific work outputs. The combined gas turbine-steam cycle burning semi-clean fuel was shown to have one of the lowest cost per unit of electricity when compared to several other methods of generating electricity in (1)**. The analyses in (2) and (3) show the potential benefits of using water cooled turbines in electric utility applications. The combined gas turbine-steam cycles studied in the references used water cooled gas turbines and had higher thermal efficiencies and specific power outputs than those with

current or advanced air-cooled gas turbines. The resultant potential benefits are conservation of fuel usage, greater flexibility in the fuel used, and reductions in the cost of electricity. Improvements in fabrication technology and non-destructive inspection techniques have added to the interest.

As a consequence of this renewed interest, a review of research related to liquid cooling of gas turbines has been conducted. Methods and information reviewed are more applicable to ground-based turbines such as those intended for electric utility peaking or combined cycle use. This is because engine weight and design considerations to incorporate liquid cooling into ground-based applications are less critical than in aircraft applications. The information contained in this report, however, is generally applicable to other gas turbines. The report reviews and discusses liquid cooling methods and systems for turbines and results from demonstrator-turbines that have been built and tested. An assessment of the state-of-the-art and future requirements has been made. Although personal inquiries on research and technology were made as late as mid-year 1977, the survey herein includes only available information published prior to that time.

REVIEW OF PAST EFFORTS

In this section liquid cooling schemes that have been tried or proposed will be reviewed. Rigs or turbines that have been built and tested to evaluate the feasibility of these schemes will be discussed concurrently. Rotor blade cooling will be emphasized because it is considered to be the most difficult problem; for completeness, liquid cooled stator tests will be discussed.

FORCED CONVECTION - Simple forced convection is the first method to be

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discussed and is shown schematically in figure 1. Water is taken on board the rotating turbine wheel through a seal, moves radially outward along the disk, and then circulates through cooling passages in the blade. The coolant then moves radially inward along the disk and exits through another seal. In this case "forced convection" is actually a misnomer; because of the large body force acting on the density difference between fluid in the cold, inlet tube and the hot, outlet tube the flow is driven by natural convection. Theoretically most liquids can be considered as coolants with this system; however, water or steam are probably the only practical choices due to possible leakage and quantities required. The main advantage of the forced convection cooling scheme is its ability to obtain lower blade temperatures than any of the other liquid cooling systems. Some other advantages of this system are simplicity and easy recovery of spent coolant thus making possible the recovery of waste heat. The main disadvantage of this system is the high coolant pressure within the blade due to the rotation if dense liquids such as water are used. For a typical, large, ground-based, electric utility gas turbine with a diameter of 2.07 meters (6.8 ft) turning at 3600 RPM the pressure at the tip of the blade would be over 75.8 m-Pascals (11 000 psi). Another disadvantage of this scheme is that foreign matter in the coolant could be centrifuged to the blade tips and block cooling passages.

Several liquid cooling demonstrators have used forced convection systems. An aluminum forced convection water cooled turbine blade was tested at the NACA Lewis Flight Propulsion Laboratory. This work is described by Kottas and Shefflin (4), Bartoo and Freche (5), Freche and Diaguila (6), and Ellerbrock (7). In operation, water was pumped out two radial passages nearest the leading edge and near the blade tip drilled cross-over passages conducted the water to two radial trailing edge passages. The spent cooling water was taken off the rotor through axial discharge holes in the disk near the blade hub. Blade temperatures were kept extremely low; the highest measured temperature was 470 K (386° F). The maximum turbine inlet gas temperature was 1422 K (2100° F). The small number of relatively large cooling passages led to high thermal gradients.

A forced convection cooling scheme similar to the schematic shown in figure 1 was used in an engine built by the Solar Aircraft Company. This work is described by Alpert et al. (8), Alpert (9), and Alpert et al. (10). The blade was made from a low alloy steel with drilled radial cooling passages which were sealed at the tip by a welded tip cap. Temperatures in

highly stressed regions were maintained below 700 K (800° F) with peak temperatures less than 867 K (1100° F) while cooling water was kept less than 367 K (200° F). Turbine inlet temperature was 1228 K (1750° F) at 4.87:1 overall pressure ratio. The large amount of heat removed from the cycle by the cooling water lowered efficiency by an estimated 20 percent. The water cooling scheme was successfully demonstrated but blade tip cap failures indicated improved fabrication and inspection techniques were required.

A variation of the forced convection cooling scheme is shown in figure 2, this is forced convection with tip ejection and collection. Coolant is taken on board the wheel through a seal or other means and after cooling the blade is ejected at the tip. Some of the water in the blade can be allowed to vaporize thus taking advantage of the heat of vaporization to lower coolant flow requirements. At the stationary shroud a collector catches the unvaporized water that is slung from the blade tip. The ejected steam enters the gas stream. Advantages of this system are the low coolant passage pressure, lower potential for cooling passage clogging because of tip ejection and lower water consumption. Other advantages are potential recovery of waste heat from the collected coolant, and the elimination of one of the rotary seals. Disadvantages include: erosion of stationary shroud from water droplets, work required to pump coolant from a low to a high radius, erosion of downstream stages from uncollected water drops, and lowering of the gas temperature entering downstream stages due to mixing of steam and uncollected water. The most difficult problem of these may be erosion by high velocity water droplets.

This cooling concept was used by General Electric and is described by Anon. (3), Kydd and Day (11), and Perugi (12). This system was designed to be used in a combined gas turbine-steam cycle for generating electricity from coal derived fuels. Cooling water is sprayed from stationary nozzles into a rotating gutter at the blade base, this eliminates the need for a rotary seal. From the rotating gutter the cooling water is routed into the cooling passages in the blade. Cooling water flow is controlled by a system of weirs at the base of the blade; this is described in detail by Kydd (13), Moore (14) and Grondahl and Eskesen (15). From the blade cooling passages water then enters a manifold and exits at the blade tip through a nozzle. The flow rate is low enough that some of the water evaporates and enters the gas stream as steam. Some of the remaining water is supposed to be caught and collected by slot in the stationary casing. This

cooling concept, minus the collection device, has been successfully tested to temperatures as high as 1830 K (2850° F) at 15 atmospheres in a 24.7 cm (9.7 inch) diameter demonstrator turbine. Limited basic heat-transfer and erosion research aimed at solving the inherent problems in this concept is being conducted at General Electric. Some of these results are reported in (16).

THERMOSYPHON - The thermosyphon is a device in which heat is transported by a moving fluid, the fluid motion being caused by a density difference in a body force field. In the case of the turbine wheel, the body force field is the centrifugal force field caused by rotation, and is proportional to the product of the radius and the square of rotational speed. This can be as high as 20,000 g's for a typical turbine; thus, there is a large potential for natural convection heat transfer. The thermosyphon can be classified into three basic types: the closed thermosyphon, the open thermosyphon, and the closed loop thermosyphon.

Closed Thermosyphon - The closed thermosyphon is illustrated in figure 3(a) in a turbine application. The thermosyphons form the cooling passages in the blade, each cooling passage is a closed tube with the primary heat transfer media (liquid or gas or both) sealed inside. Primary coolant near the tube wall is heated in the blade cooling passage. This hot, less dense, coolant then flows inward toward the cool end being displaced by cold, more dense fluid from the heat exchanger. This process is illustrated in figure 3(b). In the heat exchanger end of the tube, heat is removed from the primary fluid near the tube wall. This cool, more dense fluid moves outward toward the heated end. In the region between the heat exchanger and blade cooling passage hot and cold fluid interact. Hot fluid flowing in annular type flow in the cooling passage must find its way to the central core in the heat exchanger end and vice versa for the cold fluid in the heat exchanger. Several possible mechanisms for this transfer exist but turbulent mixing is probably the dominant mode. A secondary coolant removes heat from the heat exchanger and carries it away. The secondary coolant could be water that is externally supplied, the fuel used for combustion, or it could be compressor discharge air.

Another type of closed thermosyphon is the two phase thermosyphon or vapor chamber. The construction is similar to the ordinary closed thermosyphon but the chamber is only partially filled with working fluid. At the design condition the working fluid is vaporized in the blade cooling passage and recondenses at the cooled, heat-exchanger end.

Condensate runs outward in a film from the condenser end toward the heated end thus cooling the blade.

The closed thermosyphon has the advantage of being completely sealed thus clogging by foreign matter is not a problem, also coolant pressure within the blades can be maintained lower than for some other systems. The choice of primary working fluid is not limited as with other systems, thus liquid metals or boiling and condensing liquids with their excellent heat transfer characteristics can be considered. On the negative side, should a crack in the blade material develop, primary coolant can be lost and cooling will no longer take place. The closed thermosyphon system requires a primary to secondary coolant heat exchanger. It may be difficult to find adequate space on the turbine wheel for this and it may also be costly. If the working fluid and the thermosyphon wall material are not compatible, corrosion could cause clogging or impede efficiency for long term operation.

In the early 50's General Electric demonstrated a liquid metal cooled turbine rotor as reported by Saciu (17). Not much information is available about this work. All that is known is that the blade was run at gas temperatures over 1478 K (2200° F) and was cooled by closed thermosyphons partially filled with liquid metal. Heat was extracted at the blade base by an air heat exchanger. It was reported that heat flux at the leading edge was increased by a factor of 4 over air cooling.

Genot and LeGrives (18), and LeGrives and Genot (19) found similar results in a rotating rig with a liquid metal closed thermosyphon and concluded that the closed thermosyphon could compete with film cooling as a viable cooling scheme for aircraft engines.

Open Thermosyphon - The open thermosyphon system is shown in figure 4(a) while figure 4(b) shows a cross-section of a single cooling passage. Coolant enters a cavity or reservoir at the base of the blade. Cooling passages in the blade are open to this reservoir and the fresh, more dense coolant in the reservoir displaces the hot fluid in the cooling passages. The hot fluid from the blade then mixes with the fluid in the reservoir which is continually drained off and replaced with cool fluid from the inlet.

Advantages of this system are its simplicity and that no heat exchanger is required. Fluid pressure at the blade tip is a function of the location of the free surface in the reservoir thus pressures within the blade can be lower than for other systems. A disadvantage of the open thermosyphon is that foreign matter in the coolant can collect at the blade tips because the cooling passages are blind

holes. If the length-to-diameter-ratio of the thermosyphon is too large performance will suffer due to the growth and mixing of the boundary layers of hot and cold fluids flowing in opposite directions.

An open thermosyphon cooled turbine was designed by Schmidt during World War II in Germany. Test results on this turbine are reported by Smith and Pearson (20). The turbine operated for about 50 hours at 1367 K (2000° F) and achieved 2 hours running at 1533 K (2300° F). The inlet pressure was only 1.2 atmospheres which meant a relatively low heat load compared to a modern turbine. A four stage turbine fabricated in Germany was taken to England and tested after the war. Results of the test are given by Robinson (21) and the operation is described by Schmidt (22). Several problems developed during testing. The drum and blades were machined from low alloy steel and were subjected to corrosion from the water coolant. When testing was completed several cooling passages had been corroded completely through and most of the cooling passages were blocked with iron oxide. The turbine was tested to 923 K (1202° F) but was only able to achieve 61 percent of design speed due to vibration. The source of vibration is not known but was speculated to be caused by boiling in the cooling passages or instabilities in the water ring which formed the reservoir at the blade base.

An open thermosyphon, water cooled turbine blade was tested at the NACA Lewis Flight Propulsion Laboratory. Results of the work are reported by Bartoo and Freche (5), Freche and Diaguilia (6), Ellerbrock (7), Freche (23), Freche and Schum (24), Curren and Zalabak (25). The maximum gas temperature of 1222 K (1740° F) resulted in an average blade temperature of 450 K (350° F) and a maximum measured blade blade temperature of 739 K (870° F) at the trailing edge. It was found that increasing coolant flow could lower average blade temperature but not maximum temperature. An improvement in cooling effectiveness of passages with large length-to-diameter-ratio was obtained when they were connected by tip cross-over passages to cooling passages with smaller length-to-diameter-ratio as reported by Zalabak and Curren (26). This arrangement let cool fluid from the larger diameter passage flow unidirectionally down the small passage; the problem of hot and cold fluid flowing in opposite directions and mixing in long thin passages was relieved in this manner. This arrangement might be called an open loop thermosyphon. The open loop thermosyphon seems to be the most practical of the open thermosyphon configurations; however, no other known research has been conducted.

Closed Loop Thermosyphon - The third

type of thermosyphon is the closed loop thermosyphon shown on figure 5(a) in a possible turbine blade cooling configuration. Details of the blade cooling passages and heat exchanger are depicted on figure 5(b). Instead of coolant flowing in two directions simultaneously in the cooling passages as in the closed or open thermosyphons, flow is in one direction in each passage. Cool, more dense fluid moves radially outward through the large central feed passages and is then manifolded at the tip to flow radially inward through the small cooling passages around the perimeter of the blade. Heated fluid in the cooling passages then moves inward to a heat exchanger and is cooled by a secondary coolant.

The closed loop thermosyphon has the same advantages and disadvantages as the closed thermosyphon. However, because flow is unidirectional in all the passages, much longer passages can be utilized. Long, thin passages may be necessary in the turbine application to minimize thermal gradients.

A closed loop thermosyphon cooling scheme was demonstrated in an engine designed to run at 1533 K (2300° F) by the Continental Aviation and Engineering Corporation. The work is described by Gabel (27), Gabel and Rabbey (28), and Johnson (29). The primary coolant used was steam and the secondary coolant was fuel. The heat removed from the blades was thus added back to the cycle through the fuel. Several hundred hours of elevated temperature testing were accomplished with a maximum temperature of 1617 K (2450° F). Twenty-one hours were accumulated above 1422 K (2100° F). When the gas temperature was 1417 K (2090° F), blade temperatures were measured to be between 850 K (1070° F) and 1072 K (1470° F). Several blade failures were encountered; the cause of the blade failures was believed to be loss of primary coolant through cracks in blades or welds. Better non-destructive testing procedures might have eliminated this problem.

OTHER METHODS OF LIQUID COOLING - A different approach to using liquid to allow operation at higher turbine inlet temperatures is to lower the temperature of the compressor discharge air used for turbine cooling. Figure 6 shows two proposed methods of accomplishing this. In figure 6(a) compressor discharge air used for cooling hot section components is passed through a heat exchanger where it gives up heat to a secondary liquid coolant. The cooled cooling air then goes to conventional air cooled vanes and blades. Advantages of this method are that state-of-the-art hardware can be used, high liquid coolant pressures and the potential for erosion by liquid are

avoided. It may be feasible to use fuel to cool the cooling air thus the heat removed from the cooling air could be added back to the gas turbine cycle directly. If other liquids are used it is possible that heat removed from the cooling air by the liquid could be used elsewhere in the cycle. The main disadvantage is the added complexity and cost of the heat exchanger.

Edwards (30) described the system shown in figure 6(b). In this system liquid water is sprayed directly into the hot cooling air; the water vaporizes and cools the cooling air. Edwards states that by injecting 10 percent water into the cooling air the temperature could be reduced from 700 K (800° F) to 440 K (332° F). This could allow turbine inlet temperature to be increased by about 100 K (180 F). The obvious advantage of this system is the elimination of a costly heat exchanger. A disadvantage is that minerals in the water could clog small cooling passages or film cooling holes.

Goodyear and Waterston (31) proposed the addition of water mist to cooling air to increase its cooling capacity as well as lower its temperature. They measured heat transfer coefficients of an impinging air/water mist. The addition of 6 percent water by weight doubled the heat transfer coefficient of an impinging air jet.

The spray cooling scheme was investigated and reported by Freche and McKinnon (32), Freche and McKinnon (33), Jurke and Kemeny (34), Freche and Hickel (35), and McKinnon and Freche (36). In this scheme, water is sprayed from stationary nozzles or nozzles in the rotating blade bases onto the exterior of the rotating blades; the evaporation of water cools the blade surface. The cyclic application of the water spray cooling technique caused thermal shock failure of the blades. The large quantities of water required make the system impractical for steady state use.

"Sweat cooling" utilizes a liquid coolant that seeps out through pores in the blade wall and evaporates from the blade surface. In order to control coolant flow distribution and flow rates, very fine pores would be required in the blade. These would be subjected to clogging from foreign matter in the coolant as well as plugging from corrosion. Sweat cooling has not been tried in a turbine application.

STATOR COOLING - Tests of a water cooled stator are reported by Freche and Diaguila (6). The cooling scheme consisted of five radial holes along the camber line. The only problem was that of high trailing edge temperatures. The large spacing of the coolant holes was probably the cause of these high local temperatures.

May (37) and (38) used a static cascade to test a forced convection cooled rotor blade. The cooling scheme consisted of 4 drilled holes connected by a manifold at the blade tip. An organic fluid was used as the coolant; this allowed operation at coolant temperatures of up to 673 K (752° F) at 1.11 m-pascals (161 psi) without boiling in the passage. The maximum turbine inlet temperature tested was 1573 K (2372° F). The maximum measured blade temperature was 1013 K (1364° F) at the trailing edge. Thermal gradients were very high - this was again due to the large spacing of the cooling holes.

Stappenbeck and Moskowitz (39) tested liquid metal cooled stator vanes. An air-liquid metal heat exchanger was included in the coolant loop to simulate recuperative heating of compressor discharge air. A maximum gas temperature of 1894 K (2950° F) was achieved with no failures; however, total test duration was short. Current technology liquid metal pumps were found to be heavy and cumbersome.

TECHNOLOGY REQUIRED

Before the design and optimization of liquid cooled turbines can become a state-of-the-art process, a better understanding of the basic heat transfer and fluid flow mechanisms must be obtained. As seen in the previous section, there have been numerous liquid cooling demonstrations; however, these demonstrators were designed with little knowledge of the basic mechanisms involved. A review of basic heat transfer and fluid flow data that is applicable to liquid cooling of the gas turbine has been made in (40). The reference presents many of the more significant curves and correlations that summarize prediction capabilities, trends and technology available. This review of liquid cooling literature has pointed out the lack of basic data and understanding of heat transfer, pressure drop, critical heat flux, and flow stability in the cooling passages of a rotating turbine blade. In this section, areas of research required to advance our understanding of these processes are outlined.

Additional experimental heat transfer and flow data must be obtained and correlations and prediction methods developed for single and two phase flow particularly under high centrifugal accelerations. Some two phase heat transfer data on stationary tubes, limited pool boiling data to about 475-g's and single phase flows in a rotating tube up to 5000-g's are available; however, it is not known if these data can be

successfully applied to rotating systems where centrifugal accelerations may reach 20,000-g's and where large Coriolis forces are also present. There are no data for flow toward the axis of rotation, only for radial outflow. Information is also needed on the effect of non-uniform passage heating and high centrifugal accelerations on critical heat flux. The information that is available is sparse and only for non-rotating conditions. Pool boiling critical flu. data are available for accelerations to 5000-g's. No forced convection boiling critical flux data are available at high accelerations or with Coriolis forces. The effect of large accelerations on flow stability in boiling channels is also unknown. Research must also be conducted to determine the distribution of coolant phases in liquid-vapor systems. Centrifugal and Coriolis forces may cause some of the cooling passages to become liquid starved thus causing local overheating of the turbine blade. Information must also be obtained on the effect of variables such as coolant viscosity, surface tension and passage diameter on the wetting of the passage periphery under the influence of Coriolis forces and their combined effect on flow and heat transfer.

Heat transfer to supercritical water must be investigated under the influence of high-g forces. Correlations are available for heat transfer to supercritical water in stationary tubes but it is not known if these correlations will be valid in a high-g environment. The problems of flow stability of near-critical fluids in rotating passages must be explored experimentally for expected turbine operating conditions. There is no known analysis capable of modeling all aspects of supercritical heat transfer in the static case let alone the rotating case.

The closed loop thermosyphon or the open thermosyphon with passages connected together at the blade tip seem the most practical among the various thermosyphons for turbine applications because it permits the use of large passage length-to-diameter-ratios required to minimize stresses. The "mixed convection" analysis, where heat transfer and friction pressure drops are evaluated from forced convection correlations for a free convection driven flow, needs to be verified by experiment. The possibility of flow oscillation in the closed loop system and the consequences to heat transfer must be investigated.

Other areas related to liquid cooling where technology needs to be broadened include experimental confirmation of the expected influence of highly cooled walls on combustion gas stream boundary layer development and heat transfer. A

currently available two-dimensional boundary layer program predicts early transition to turbulent flow and thus higher heat transfer for highly cooled walls. This effect could cause higher metal temperatures than expected if not accounted for in a design and should be investigated experimentally. The clogging of small cooling passages by deposits from cooling water needs to be evaluated as well as methods of minimizing erosion of cooling passages and water recovery systems. Long life seals to get coolant on and off rotating turbine wheels are needed. Improved manufacturing and non-destructive inspection techniques need to be developed that allow the economical production of blades, vanes and associated hardware that will be capable of containing coolant at the high pressures required for the long lifetimes of large industrial gas turbines.

CONCLUDING REMARKS

Although a number of liquid-cooled turbine demonstrators have been tested and some basic and supportive liquid-cooling research has been conducted, existing technology is not adequate to design a liquid-cooled turbine with a high assurance of success or to optimize liquid cooling systems for a given turbine application. Considerable basic and applied research on liquid cooling of gas turbines is still required.

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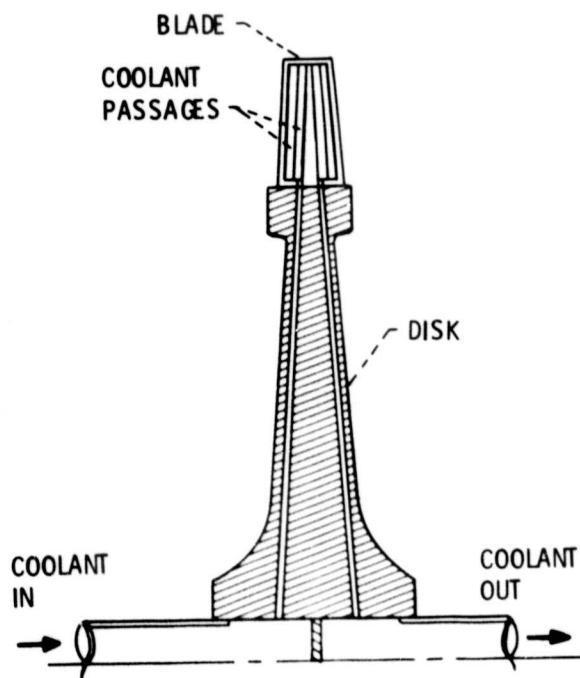


Fig. 1 - Simple forced convection

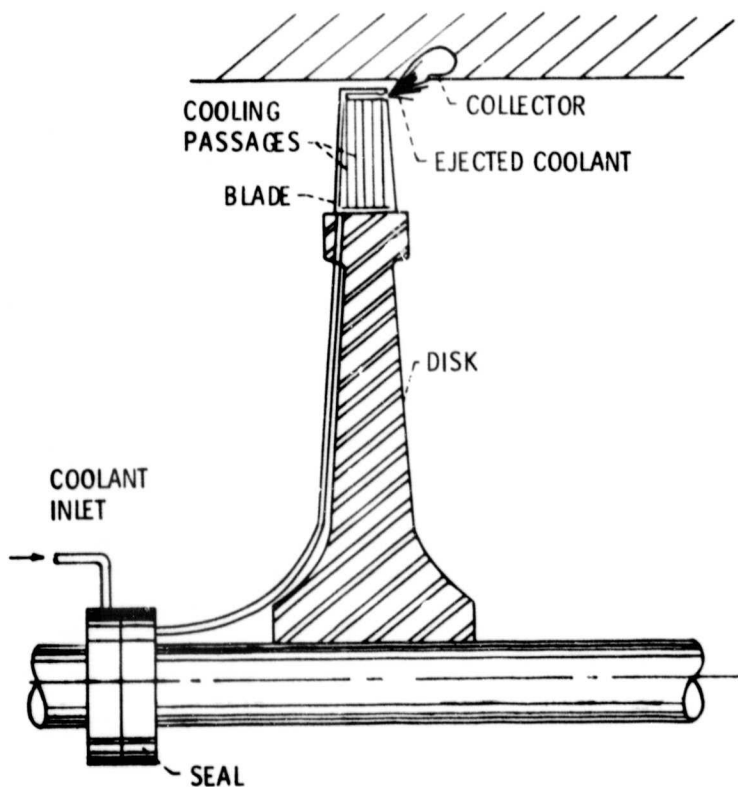
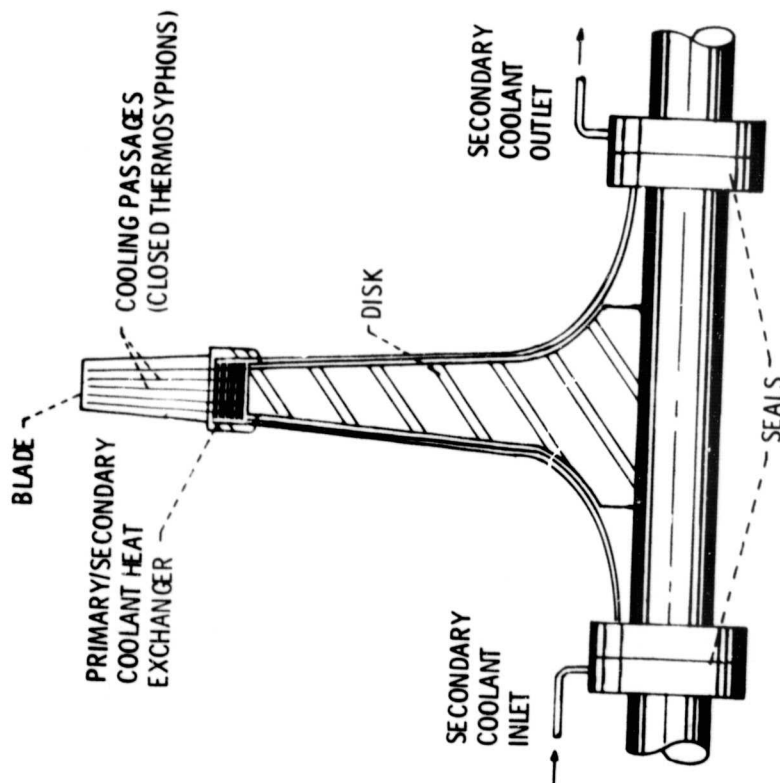
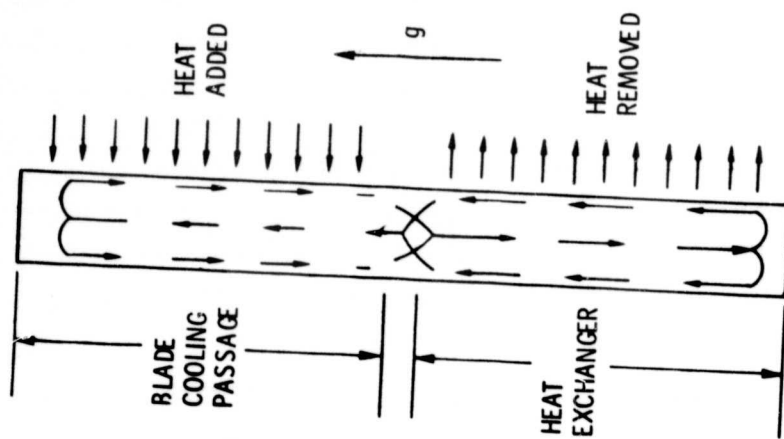


Fig. 2 - Forced convection with tip ejection and collection

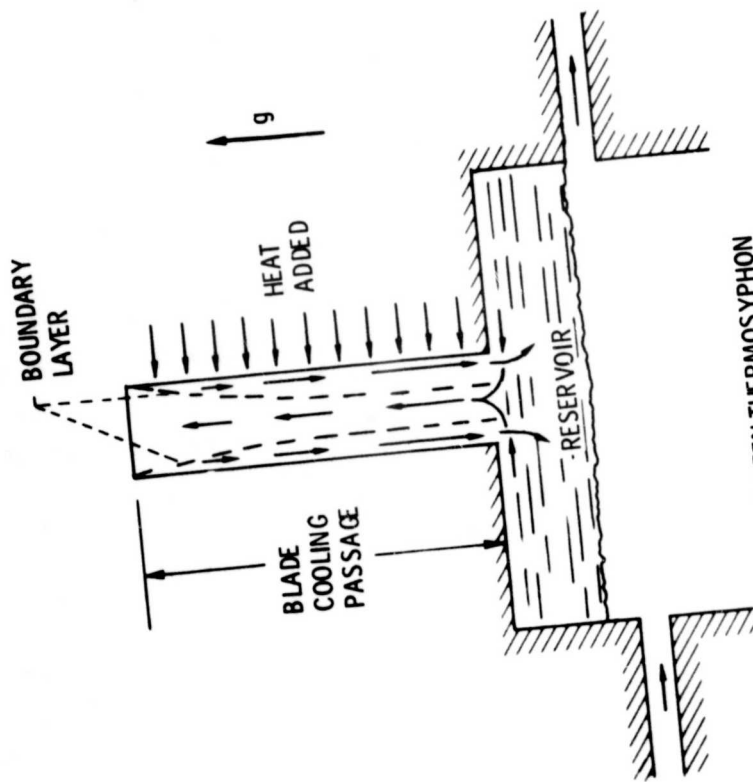
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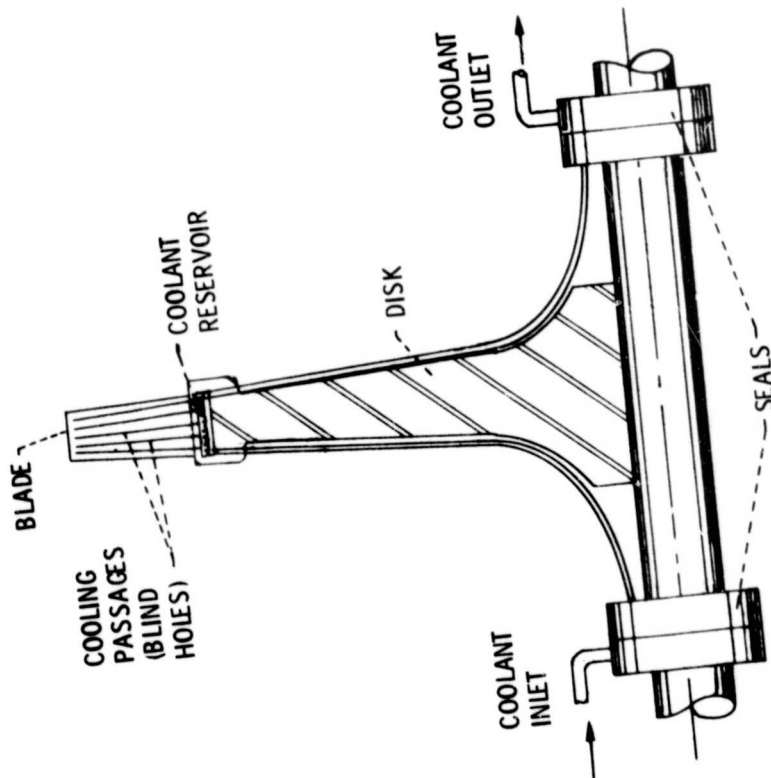
(a) POSSIBLE GAS TURBINE APPLICATION
Fig. 3 - The closed thermosyphon system



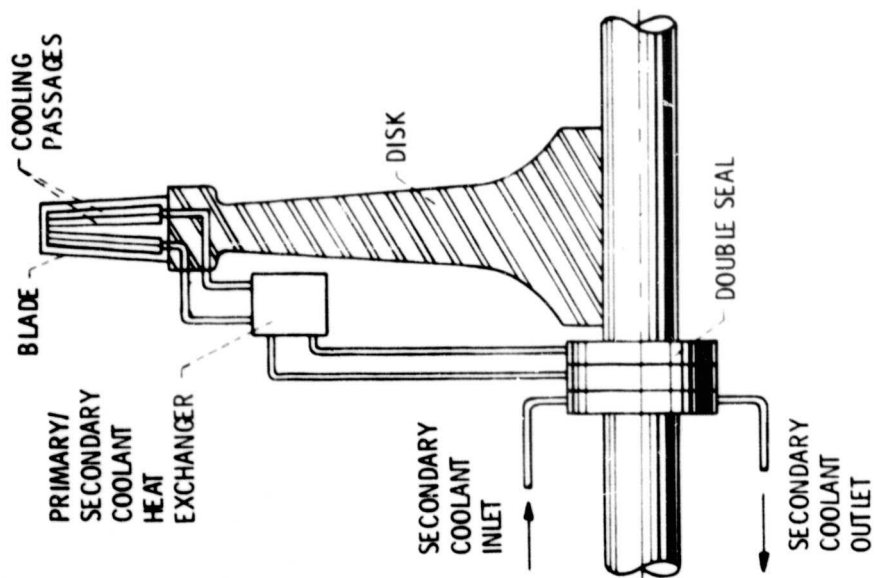
(b) CLOSED THERMOSYPHON TUBE
Fig. 3 - Concluded



(b) OPEN THERMOSYPHON
Fig. 4 - Concluded

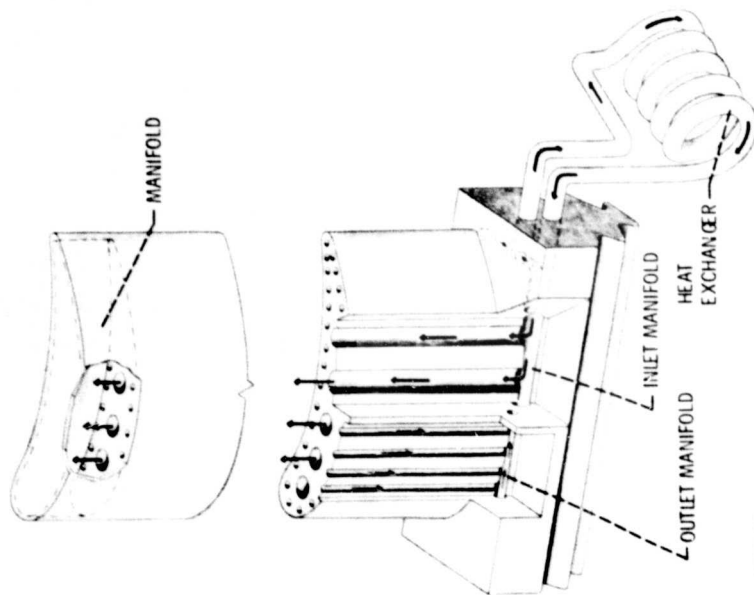


(a) POSSIBLE TURBINE COOLING CONFIGURATION
Fig. 4 - Open thermosyphon system



(a) POSSIBLE TURBINE COOLING CONFIGURATION

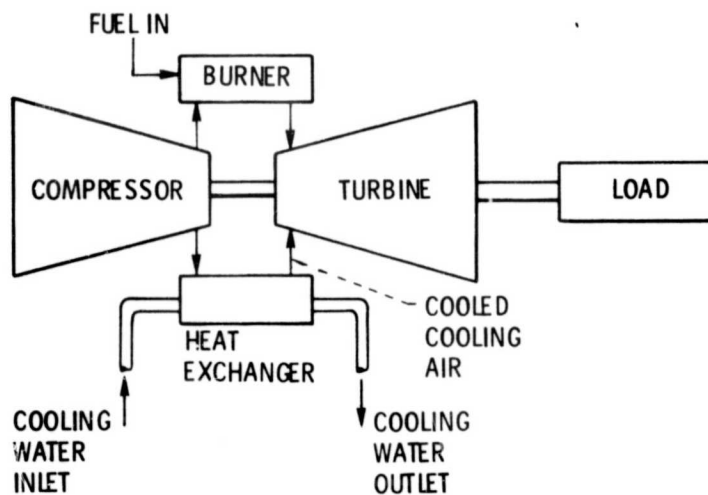
Fig. 5 - Closed loop thermosyphon system



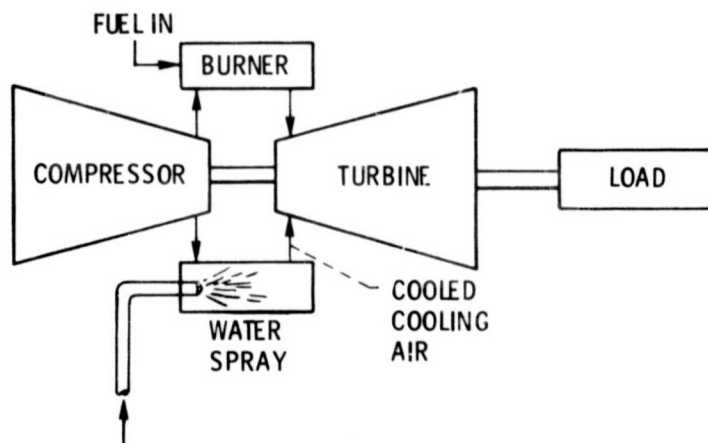
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(b) DETAILS OF BLADE COOLING PASSAGES AND HEAT EXCHANGER

Fig. 5 - Concluded



(a) AIR/WATER HEAT EXCHANGER SYSTEM



(b) WATER SPRAY SYTEM

Fig. 6 - Cooled cooling air configurations

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